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A Two-Stroke Diesel Engine Simulation Program

Jon H. VanGerpen
Engine Research Institute
Iowa State University
Ames, Iowa

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A TWO-STROKE DIESEL ENGINE SIMULATION PROGRAM

Jon H. Van Gerpen
Engine Research Institute
Iowa State University
Ames, Iowa

SUMMARY

A computer program simulating a two-stroke diesel engine is developed and documented. The program is suitable for simulating the diesel core of a high-output combined-cycle diesel engine. The engine cylinder and the intake and exhaust ports are defined as independent thermodynamic systems and the mass and energy equations for these systems are developed. A single zone combustion model is used and perfect mixing during scavenging is assumed. The program input requirements and output results are discussed. A sample case is provided for an opposed piston, uniflow scavenged two-stroke diesel engine.

E-5121

INTRODUCTION

This report documents a computer program solving the equations from a mathematical model that simulates a two-stroke diesel engine. The model considers the thermodynamics and fluid mechanics of the working fluid from the entrance of the intake port to the exit of the exhaust port. The program can predict the effect on engine performance of changes in parameters such as speed, boost pressure, valve timing, and fueling level. The program output provides information about power output, brake mean effective pressure (BMEP), heat transfer losses, and cylinder pressures and temperatures. The program was written in modular form so that the submodels could be modified or replaced without requiring program alteration.

Most of the input data is supplied by an input data set, but because the program is intended to be used as part of a larger program that will match the diesel with attached turbomachinery, the inlet and exit conditions and the mass flow rate are communicated through a subroutine argument list.

This report discusses the mathematical model used in the program and describes the subroutines that make up the model. Input requirements are stated and the output listing is explained. A sample program run is also provided.

SYMBOLS

ATDC after top dead center

C_1 shape parameter for diffusion burning

C_2 crank angle for start of combustion

C_3 combustion duration parameter

C_4	premixed burning fraction
C_5	shape factor for premixed burning
F_s	stoichiometric fuel-to-air ratio
f	residual fraction
h	enthalpy
h_{E_1}	enthalpy of gases exiting to exhaust manifold
h_{E_2}	enthalpy of gases exiting to exhaust port
h_{I_1}	enthalpy of gases entering intake port
h_{I_2}	enthalpy of gases entering cylinder
m	mass
m_{cyl}	mass of cylinder gases
$\dot{m}_{E_1}$	mass flow rate of gases from exhaust port to exhaust manifold
$\dot{m}_{E_2}$	mass flow rate of gases from cylinder to exhaust port
m_{FC}	mass of fresh charge (pure air)
$\dot{m}_f$	mass flow rate of fuel added to cylinder
$m_{f,tot}$	total amount of fuel added to cylinder
$m_{fuel,new}$	mass of new fuel injected
m_{IP}	mass at intake port
$\dot{m}_{I_1}$	mass flow rate of gases from intake manifold to intake port
$\dot{m}_{I_2}$	mass flow rate of gases from intake port to cylinder
$m_{R,BA}$	mass of burned air in residual
$m_{R,fuel}$	mass of burned fuel in residual
$m_{R,UA}$	mass of unburned air in residual
P	purity
p	pressure
$\dot{Q}$	heat transfer rate

R	gas constant
R_{IP}	gas constant at intake port
r_D	delivery ratio
s	entropy
T	temperature
T_{IP}	temperature at intake port
U_{cyl}	internal energy of the cylinder gases
U_{EP}	internal energy of the exhaust port gases
U_{IP}	internal energy of the intake port gases
u	internal energy per unit mass
u_{cyl}	internal energy of the cylinder gases per unit mass
V	volume
W	rate of doing work, also equal to $p\ dV/d\theta$
y	nondimensional time parameter
ϕ_{cyl}	equivalence ratio of cylinder gases
ϕ_{E_1}	equivalence ratio of mass leaving exhaust port to exhaust manifold
ϕ_{E_2}	equivalence ratio of mass leaving cylinder to exhaust port
ϕ_{I_2}	equivalence ratio of mass entering cylinder from intake port
θ	crank position, or crank angle
η_{ch}	charging efficiency
η_{sc}	scavenging efficiency
η_{tr}	trapping efficiency

THE MATHEMATICAL MODEL

The mathematical model for the engine is based on applying the first law of thermodynamics to the systems composed of the engine cylinder and the intake and exhaust ports. The ideal gas equation and equilibrium gas properties are also used to solve for the pressure and temperature in the cylinder.

Figure 1 shows the systems being analyzed in this section. The first system, the engine cylinder, receives mass from the intake port system and delivers mass to the exhaust port system. The possibility for backflow exists when the cylinder pressure is above the intake port pressure or below the exhaust port pressure. The first law of thermodynamics can be written as follows for the system consisting of the engine cylinder:

$$\frac{dU_{cyl}}{d\theta} = \dot{Q} - \dot{W} + \dot{m}_{I_2} h_{I_2} + \dot{m}_f h_f - \dot{m}_{E_2} h_{E_2} \tag{1}$$

where

U_{cyl} the internal energy of the cylinder gases

$\dot{Q}$ the heat transfer rate from the cylinder

$\dot{W}$ the rate at which work is done by the cylinder gases

$\dot{m}_{I_2}$ the mass flow rate into the cylinder from the intake port

$\dot{m}_{E_2}$ the mass flow rate from the cylinder to the exhaust port

$\dot{m}_f$ the mass flow rate of fuel added to the cylinder

h_{I_2} enthalpy of gases entering from intake port

h_f enthalpy of fuel added to the cylinder

h_{E_2} enthalpy of gases exiting cylinder to exhaust port

θ crank position, or crank angle (used as a time parameter)

The derivative of U_{cyl} can be written as

$$\frac{dU_{cyl}}{d\theta} = \frac{d(m_{cyl} u_{cyl})}{d\theta} = m_{cyl} \frac{d(u_{cyl})}{d\theta} + u_{cyl} \frac{dm_{cyl}}{d\theta} \tag{2}$$

where m_{cyl} is the mass of cylinder gases and u_{cyl} is the specific internal energy of the cylinder gases.

The ideal gas equation can be differentiated with respect to crank angle to give the following expression:

$$p \frac{dV}{d\theta} + V \frac{dp}{d\theta} = mR \frac{dT}{d\theta} + mT \frac{dR}{d\theta} + RT \frac{dm}{d\theta} \tag{3}$$

Each of the properties p, V, m, R, and T refer to the cylinder gases.

Derivatives of u and R with respect to θ can be expressed in terms of derivatives of T, P, and ϕ, the equivalence ratio, as shown in equations (4) and (5). The equivalence ratio used here is defined as the actual fuel-to-air mass ratio divided by the stoichiometric fuel-to-air ratio.

$$\frac{dR}{d\theta} = \frac{\partial R}{\partial p}\frac{dp}{d\theta} + \frac{\partial R}{\partial T}\frac{dT}{d\theta} + \frac{\partial R}{\partial \phi}\frac{d\phi}{d\theta} \tag{4}$$

$$\frac{du}{d\theta} = \frac{\partial u}{\partial p}\frac{dp}{d\theta} + \frac{\partial u}{\partial T}\frac{dT}{d\theta} + \frac{\partial u}{\partial \phi}\frac{d\phi}{d\theta} \tag{5}$$

Equations (3) and (4) can be combined and solved for $dT/d\theta$. This equation can then be used to eliminate $dT/d\theta$ in the equation that results from combining equations (1) and (5) to obtain an expression for $dp/d\theta$.

The equivalence ratio in the cylinder changes because of the addition of air from the intake port, fuel from the fuel injector, or backflow from the exhaust port. The following expression can be obtained for the variation of ϕ_{cyl} with crank angle:

$$\frac{d\phi_{cyl}}{d\theta} = \frac{1 + \phi_{cyl}F_s}{m_{cyl}}\left(\frac{\phi_{I_2} - \phi_{cyl}}{1 + \phi_{I_2}F_s}\dot{m}_{I_2} - \frac{\phi_{E_2} - \phi_{cyl}}{1 + \phi_{E_2}F_s}\dot{m}_{E_2} + \frac{\dot{m}_f}{F_s}\right) \tag{6}$$

where ϕ_{cyl}, $\dot{m}_{I_2}$, $\dot{m}_{E_2}$, and $\dot{m}_f$ are defined previously and

where

ϕ_{I_2} equivalence ratio of mass entering or leaving cylinder through the intake opening

ϕ_{E_2} equivalence ratio of mass entering or leaving cylinder through the exhaust opening

F_s stoichiometric fuel-to-air ratio

The equations described previously for $dp/d\theta$, $dT/d\theta$, and $d\phi/d\theta$ can be integrated simultaneously to obtain the p, T, and ϕ in the cylinder as a function of θ. The heat transfer rate is calculated with Annand's correlation (ref. 1). The mass entering and leaving the cylinder through the intake and exhaust openings is calculated on the basis of an isentropic expansion into a region of lower pressure. Deviations from this ideal flow are incorporated through the use of a user-supplied flow coefficient. The mass flow rate of fuel added to the cylinder is also user-specified through a functional expression given in subroutine DWCA.

This model assumes that the mass leaving a system has the same properties as the system. This assumption implies that there will be perfect and instantaneous mixing between any mass added to the system and the mass already existing in the system. This has implications for the scavenging process because the actual scavenging process can be more or less effective than perfect mixing. If there is considerable short circuiting of the intake air directly to the exhaust port, then the actual scavenging efficiency will be less than calculated here, but, if the incoming air is able to push the exhaust products out of the cylinder without excessive mixing, then the actual scavenging efficiency will be higher. The actual scavenging efficiency in an engine is highly sensitive to combustion chamber design and engine operating conditions, so the perfect mixing assumption represents a reasonable compromise that should still allow the prediction of trends.

The second two systems identified in figure 1, the intake port and the exhaust port are described by equations similar to those developed previously. The following assumptions apply to these systems:

(1) Volumes of intake and exhaust ports are constant.

(2) Pressures in intake and exhaust ports are constant and equal to the user-specified values of intake and exhaust manifold pressures.

(3) The intake and exhaust ports are adiabatic. There is no heat transfer.

(4) Mass passing from the intake manifold into the intake port is assumed to be air only. No combustion product residual gases are present. If backflow occurs from the intake port system to the intake manifold, this mass is also assumed to be air only, regardless of the actual equivalence ratio in the intake port.

(5) Mass passing from the exhaust port to the exhaust manifold has properties equal to the current values in the exhaust port. If backflow occurs from the exhaust manifold to the exhaust port, this mass also has properties equal to the current properties of the gases in the exhaust port.

The first law of thermodynamics can be written as follows for the intake port system:

$$\frac{dU_{IP}}{d\theta} = \dot{m}_{I_1} h_{I_1} - \dot{m}_{I_2} h_{I_2} \tag{7}$$

where

U_{IP} the internal energy of the intake port gases

$\dot{m}_{I_1}$ the mass flow rate into the intake port from the intake manifold

h_{I_1} the enthalpy of the gases entering the intake port from the intake manifold

The next step is to differentiate the ideal gas equation subject to the constraint that the pressure and volume are constant.

$$0 = m_{IP}R_{IP}\frac{dT_{IP}}{d\theta} + m_{IP}T_{IP}\frac{dR_{IP}}{d\theta} + R_{IP}T_{IP}\frac{dm_{IP}}{d\theta} \tag{8}$$

Equations (4) and (5) can be used to represent the derivatives of u and R with respect to θ although the pressure terms can be eliminated because of the assumption that the intake port pressure is constant.

The equivalence ratio in the intake port changes both because of the addition of air from the intake manifold and because of a possible backflow from the cylinder. The following expression can be obtained for the variation of ϕ_{IP} with crank angle:

$$\frac{d\phi_{IP}}{d\theta} = \frac{(I + \phi_{IP}F_s)}{m_{IP}}\left[\frac{\left(\phi_{IP} - \phi_{I_2}\right)}{I + \phi_{I_2}F_s}\dot{m}_{I_2} - \phi_{IP}\dot{m}_{I_1}\right] \tag{9}$$

where

ϕ_{IP} equivalence ratio of mass in intake port

In a manner similar to the case for the cylinder, equation (8) can be solved for $dT/d\theta$ to obtain the differential equation for temperature; this equation can then be used to eliminate the $dT/d\theta$ term that appears in equation (7) after substitution of equation (5). In this case, the unknown in equation (7) becomes $\dot{m}_{I_1}$. Thus, the equations for $\dot{m}_{IP}$, $dT_{IP}/d\theta$, and $d\phi_{IP}/d\theta$ can be integrated along with the equations for the in-cylinder conditions to obtain m_{IP}, T_{IP}, and ϕ_{IP}. The intake port conditions are of little interest themselves, but they are required to fix the boundary conditions of the equations describing in-cylinder conditions. Although they will not be listed here, equations can be developed for the exhaust port in exactly the same manner as for the intake port.

The equations described previously must be integrated through the engine cycle. Additional quantities, while not required to simulate the system, will also be summed during the cycle. There are a total of 17 equations. The integration process is started at the crank position where the exhaust valve opens. Initial values are required for each of the variables. While most of the variables are initialized to zero, variables such as the cylinder temperature and pressure must have initial, nonzero values. Since these values will not generally be known in advance, values are assumed and the integration is started. Since the process being described is a mechanical cycle, the properties at the end of the integration should be the same as at the start. The solution process is iterative, where the final values of the integrated variables are used as the initial values of the next iteration. When the difference between the initial and final values is less than a prescribed value, the program is considered to have converged and the program halts.

The program offers two options for integration routines. A subroutine called RKGS can be used and is supplied with the DSL2 program. However, if the IMSL library is available, the fifth-order Runge-Kutta integration routine DVERK, which can be used instead, results in much faster execution time. Statements to call each routine are provided in the program.

SUBROUTINE DESCRIPTIONS

INPUT1: This subroutine reads in the data set containing the operating parameters for the engine being simulated. A description of the input quantities is contained in the section on input data. This subroutine also does some preliminary processing of the input data. In particular, this subroutine calls the SPLINE subroutine to generate the array of spline coefficients for the intake and exhaust flow areas. The flow area information is supplied with the input data, either as a table of areas versus crank position or as port geometry, and then a cubic spline is fit to this data. This operation is only performed once and after this time, the spline coefficient array is accessed by the function subroutines AREAIN and AREAEX to calculate the intake and exhaust flow areas for any crank angle.

MODEL: This subroutine evaluates the derivatives of the differential equations that result from the mathematical model of the diesel engine. MODEL determines whether the intake and exhaust valves are open, and if so, calls TMI2 and TME2 to get the mass flow rates into and out of the cylinder. Then it calls PERX to get the properties of the gases entering and leaving the cylinder. These quantities are used in setting up the derivatives of the first-order differential equations that will be integrated by DVERK. The latest estimate of the function values enters the subroutine in the Y array and the newly evaluated function derivatives are returned in the DERY array. The program is integrating 17 simultaneous equations. The quantities being integrated are defined as follows:

Y(1)	cylinder pressure
Y(2)	cylinder temperature
Y(3)	equivalence ratio of cylinder contents
Y(4)	mass of gases in cylinder
Y(5)	cumulative mass entering cylinder
Y(6)	temperature of gases in intake port
Y(7)	equivalence ratio of gases in intake port
Y(8)	cumulative mass leaving cylinder
Y(9)	temperature of gases in exhaust port
Y(10)	equivalence ratio of gases in exhaust port

Y(11) energy entering cylinder with the intake air

Y(12) energy leaving cylinder with the exhaust products

Y(13) work done by cylinder gases

Y(14) cumulative mass of fuel injected

Y(15) cumulative heat transfer

Y(16) cumulative air entering intake port

Y(17) cumulative air leaving exhaust port

RKGS: This is a utility integration routine. It uses a fourth-order Runge-Kutta technique. RKGS calls the subroutine FCT, which in turn calls the subroutine MODEL to obtain the derivatives of the differential equations being integrated. Although the program is currently set up to use RKGS, statements are provided to change to DVERK, a routine from the IMSL library. This routine provides much faster execution times than RKGS.

SPLINE: This is a utility routine used to calculate coefficients of a cubic spline for the intake and exhaust flow areas versus crank position.

AREAIN: This is a function subroutine that, given a crank angle as an input argument, searches through an array of spline coefficients to find the appropriate set and then evaluates the spline to provide the instantaneous intake valve or port flow area.

AREAEX: This function subroutine is identical to AREAIN in that it evaluates a cubic spline to find the instantaneous exhaust valve or port flow area corresponding to a specified crank angle.

PERX: This is a utility routine used to evaluate equilibrium thermodynamic properties of combustion product gases. Given T, P, and ϕ (the equivalence ratio), this routine will compute u, h, R, s, and their derivatives with respect to T, P, and ϕ. All input and output information for this routine is passed through two COMMON blocks. Documentation of this routine is provided in reference 2.

VOL: This subroutine calculates the cylinder volume for a specified crank angle. When the subroutine argument MODE is set correctly, the subroutine can calculate the volume for a conventional piston-cylinder geometry or for an opposed piston configuration with both crankshafts in phase.

DWCA: This function subroutine is used to compute the fuel mass burning rate supplied to the single-zone first-law energy balance for the cylinder given by equation (1). According to a function developed by Watson, Pilley, and Marzouk for simulating diesel combustion (ref. 3), the mass burning rate is calculated. Although this function was developed for diesel engines operating over a narrow range of conditions, it contains five parameters that can be used to specify the shape of almost any conceivable burning-rate profile. The parameters are denoted by C_1, C_2, C_3, C_4, and C_5. The function describing the burning rate is obtained by superimposing two algebraic functions. The first function is an exponential curve given by

$$F_1 = 6.907755(C_5 + 1)y^{C_1}\exp\left(-6.907755y^{C_1+1}\right) \tag{10}$$

where

$$y = \frac{\theta - C_2}{C_3} \qquad \text{for} \qquad \theta > C_2$$

and

$$y = 0 \qquad \text{for} \qquad \theta < C_2$$

This function is intended to represent the relatively slow and controlled diffusion burning in the cylinder. A second curve, given by

$$F_2 = 5000(C_5 + 1)y^{C_5}\left(1 - y^{C_5+1}\right)^{4999} \tag{11}$$

represents rapid, uncontrolled premixed burning. These two functions are combined with a weighting factor that determines the relative amount of fuel that burns in the premixed and diffusion modes

$$\dot{m}_f = m_{f,tot}\left[C_4F_2 + (1 - C_4)F_1\right] \tag{12}$$

where $m_{f,tot}$ is the total amount of fuel added to the cylinder. Therefore, as used in these functions, the parameters C_1 and C_5 represent shape factors for the diffusion and premixed burning curves, respectively. C_2 is the crank angle at which burning is intended to start, and C_3 is proportional to the burning duration. C_4 is the weighting factor that represents the fraction of fuel burned in the premixed mode.

Q: This function subroutine calculates the heat loss from the cylinder gases to the walls during combustion. The calculation follows Annand's correlation described in reference 1. This correlation has the feature that it treats the radiation and convection losses separately. The cylinder walls are divided into three areas for the heat transfer calculations: the head, the piston, and the cylinder bore or sleeve. The head and piston areas are constant but the sleeve area changes as the piston moves up and down. The wall temperature data for each of these areas are entered by the user through the variables THEAD, TPISTN, and TSLEEV in the input data set. While the variation with time of the in-cylinder heat transfer is calculated using a heat transfer coefficient determined from Annand's correlation, the total amount of heat loss from the cylinder will normally need to be adjusted, or tuned, by the user to give the expected fraction of fuel energy lost to heat transfer. This is done by varying the ANNND parameter in the input data set. When set equal to 1.0, the heat transfer is calculated without modification, according to Annand's correlation. If ANNND is different than 1.0, then it is used as a multiplier to increase or decrease the in-cylinder heat transfer coefficient that correspondingly changes the amount of heat loss.

ISEN: This is a utility subroutine that uses PERX to compute the properties at the end of either an isentropic compression or expansion. The calling program provides the T, P, and ϕ of the initial state and the final pressure, and ISEN iteratively determines the temperature of the final state that has the same entropy. Although this routine uses the assumption of constant specific heats to obtain an initial estimate of the solution, the subroutine is intended for those situations where the specific heats vary significantly during the isentropic process.

TMI2: This subroutine calculates the mass flow rate through the intake valve. By checking the pressures in the intake port and the cylinder, TMI2 determines whether the flow is into or out of the cylinder. TMI2 calls ISEN to determine the properties of the downstream flow and calls AREAIN to get the instantaneous flow area. These quantities allow the determination of the ideal or isentropic flow through the valve. Then a user-supplied flow coefficient is applied to obtain an adjusted or actual flow rate into or out of the cylinder. When backflow occurs from the cylinder into the intake port, the calculated flow rate is given a minus sign to denote the mass loss from the cylinder.

TME2: This subroutine is the same as TMI2 except that the calculations are performed for the exhaust valves.

OUTPUT: This subroutine performs some of the final calculations and writes the results out both to the terminal and to an output data file. Friction is calculated using an empirically based polynomial that depends only on the mean piston speed. A friction mean effective pressure is calculated, then subtracted from the indicated mean effective pressure to give the brake mean effective pressure. This subroutine also calculates the various efficiencies used to characterize the scavenging process. The calculations are described in more detail in the section OUTPUT.

INPUT DATA

The diesel program is intended to be run as a subroutine with part of the input variables in the subroutine argument list. The subroutine argument list is as follows:

SUBROUTINE DSL2(T1,P1,P2,T2,XM)

where

T1	intake manifold temperature, °R	INPUT
P1	intake manifold pressure, psia	INPUT

P2	exhaust manifold pressure, psia	INPUT
T2	exhaust temperature, °R	OUTPUT
XM	air flow rate into engine, lbm/s	OUTPUT

The remaining input data to the diesel program are contained in a data set that is read, format free, from an input file titled DSLINP. The data must be entered as shown in table I, where the variables in the input data set are defined as follows:

RPM	diesel engine crankshaft speed, rev/min
WFCY	weight (mass) of fuel added per cycle, lbm
BORE	cylinder bore, in.
STROKE	stroke, in.
CONROD	connecting rod length, in.
CR	geometric compression ratio
MODE	piston cylinder mode (MODE = 1 for conventional piston-cylinder configuration, MODE = 2 for opposed piston configuration)
C1	shape parameter for diffusion burning
C2	crank angle for start of combustion
C3	combustion duration parameter
C4	premixed burning fraction
C5	shape parameter for premixed burning
ANNND	multiplier for Annand heat transfer correlation
THEAD	cylinder head temperature, °R
TPISTN	piston temperature, °R
TSLEEV	cylinder wall temperature, °R
MEXH	exhaust valve or port indicator (MEXH = 0 for exhaust ports, MEXH = 1 for exhaust valves)
EVO	exhaust valve or port opening crank angle, °ATDC
EVC	exhaust valve or port closing crank angle, °ATDC

MINT intake valve or port indicator (MINT = 0 for intake ports, MINT = 1 for intake valves)

AVO intake valve or port opening crank angle, °ATDC

AVC intake valve or port closing crank angle, °ATDC

NTEXH number of exhaust valve area versus crank angle data points to be read in

CDEXH exhaust valve or port flow coefficient

ALPHEX(I) array of crank angles for exhaust valve flow areas, °ATDC

FEXH(I) array of exhaust valve flow areas, $in.^2$

WIDTHE fraction of the cylinder circumference devoted to exhaust port

NTINT number of intake valve area versus crank angle data points to be read in

CDINT intake valve or port flow coefficient

ALPHIN(I) array of crank angles for intake valve flow areas, °ATDC

FINT(I) array of intake valve flow areas, $in.^2$

WIDTHI fraction of the cylinder circumference devoted to intake port

When MEXH = 1, exhaust valves are being used and the user must specify NTEXH, CDEXH, and the arrays of crank angle versus flow area. When MEXH = 0, exhaust ports are used and the user need only specify WIDTHE and CDEXH. The case illustrated previously corresponds to an engine that has exhaust valves and intake ports.

OUTPUT

The output from the program simulating a two-stroke diesel engine consists of two parts. The first part, a summary of calculated quantities that characterize the engine's performance, is discussed in detail hereinafter. The second part is a listing of the cylinder pressure and temperature, and other quantities at 5° intervals during the engine cycle. The abbreviations used to identify the columns are defined as follows:

CA crank angle at which data were calculated; top dead center corresponds to 0°

PCYL cylinder pressure

TCYL cylinder temperature. Since the program is based on a single-zone combustion model, a single temperature is defined for the entire cylinder contents.

PHICYL equivalence ratio of the cylinder gases. This is the ratio of the fuel-to-air mass ratio in the cylinder to the stoichiometric fuel-to-air mass ratio.

MCYL mass of cylinder contents

MDOTIN mass flow rate into cylinder

MDOTOUT mass flow rate leaving the cylinder

BALANCE the amount of imbalance in the energy balance. It is calculated and listed as a check on the solution of the differential equations. If changes are made to the program, such as the addition of new subroutine modules, the energy balance should be monitored to ensure that it stays close to zero.

The summary part includes a listing of the input conditions for the run and gives the engine configuration and the engine operating condition. Then a listing of calculated quantities is provided based on the program results. The first quantities calculated are the air and fuel flow rates per cylinder. The flow rates are calculated by dividing the total amount of air and fuel added to the engine over one cycle by the time required for one cycle.

The IMEP is calculated by summing the work done over the cycle and then dividing by the displacement volume. The BMEP is then calculated by subtracting the FMEP that was calculated by using an empirical correlation from the IMEP. The same process is used to obtain the indicated and brake power, the indicated and brake specific fuel consumption, and the indicated and brake thermal efficiency.

During the integration process of solving the differential equations of the mathematical model, the maximum values of cylinder temperature and pressure are recorded along with the time at which they occur. These results are also provided in the summary section.

In the literature a variety of parameters have been defined to characterize the scavenging process. These quantities can be calculated from a knowledge of how much residual gas is present in the cylinder and how much fresh charge was added. Since the instantaneous equivalence ratio in the cylinder is known from the solution of equation (6), we can determine the values of ϕ at the point where the exhaust valve opens, ϕ_{evo}, and at the point where compression starts, ϕ_{cmp}. The total mass in the cylinder is also known from solution of the cylinder mass balance. To characterize the cylinder contents, it is convenient to divide the total cylinder mass into the following five categories:

m_{FC} mass of fresh charge (pure air)

$m_{fuel,new}$ mass of new fuel injected

$m_{R,fuel}$ mass of burned fuel in residual

$m_{R,BA}$ mass of burned air in residual

$m_{R,UA}$ mass of unburned air in residual

The definitions of these quantities are based on the assumption that the equivalence ratio in the cylinder is less than 1.0. This is a good assumption for diesel engines since they normally operate with a maximum equivalence ratio of 0.6 to 0.8. The definitions also assume that the combustion reactions cause the fuel and air mixture to go to equilibrium products. These quantities can be determined by solving the following five equations:

$$\phi_{cmp}F_s = \frac{m_{R,fuel}}{m_{FC} + m_{R,BA} + m_{R,UA}} \tag{13a}$$

$$\phi_{evo}F_s = \frac{m_{R,fuel} + m_{fuel,new}}{m_{FC} + m_{R,BA} + m_{R,UA}} \tag{13b}$$

$$\frac{m_{R,fuel}}{m_{R,BA}} = F_s \tag{13c}$$

$$\phi_{evo}F_s = \frac{m_{R,fuel}}{m_{R,BA} + m_{R,UA}} \tag{13d}$$

$$m_{R,fuel} + m_{R,BA} + m_{R,UA} + m_{FC} + m_{fuel,new} = m_{cyl} \tag{13e}$$

These equations can be reduced to

$$m_{R,BA} = \frac{\phi_{cmp}}{1 + \phi_{evo}F_s} m_{cyl} \tag{14a}$$

$$m_{R,fuel} = F_s m_{R,BA} \tag{14b}$$

$$m_{R,UA} = (1/\phi_{evo} - 1)m_{R,BA} \tag{14c}$$

$$m_{FC} = (1/\phi_{cmp} - 1/\phi_{evo})m_{R,BA} \tag{14d}$$

$$m_{fuel,new} = (\phi_{evo}/\phi_{cmp} - 1)F_s m_{R,BA} \tag{14e}$$

Once the mass in the cylinder has been characterized, the various scavenging indicators can be calculated.

$$\text{purity} = P = \frac{m_{FC} + m_{R,UA}}{m_{FC} + m_{R,BA} + m_{R,UA} + m_{R,fuel}} \tag{15}$$

$$\text{residual fraction} = f = \frac{m_{R,BA} + m_{R,UA} + m_{R,fuel}}{m_{FC} + m_{R,BA} + m_{R,UA} + m_{R,fuel}} \tag{16}$$

$$\text{trapping efficiency} = \eta_{tr} = \frac{m_{FC}}{\text{total mass of air entering engine}} \tag{17}$$

$$\text{charging efficiency} = \eta_{ch} = \frac{m_{FC}}{\rho_{intake} V_{disp}} \tag{18}$$

$$\text{delivery ratio} = r_D = \frac{\eta_{ch}}{\eta_{tr}} \tag{19}$$

The scavenging efficiency, which is defined as the mass of delivered air that is retained, divided by the mass of trapped cylinder charge, is calculated using the following equation that is developed in Schweitzer (ref. 4):

$$\text{scavenging efficiency} = \eta_{sc} = \frac{1}{\left[1 + \frac{(1/P - 1)}{\phi_{evo}}\right]} \tag{20}$$

The summary also determines what happened to the fuel energy. The fraction of the fuel energy, as reflected by the lower heating value, which goes to work, to heat transfer, and to the exhaust, is calculated

$$\%\ \text{work} = \frac{W_{tot}}{m_{fuel,new} HVL} \tag{21}$$

$$\%\ \text{heat loss} = \frac{Q_{tot}}{m_{fuel,new} HVL} \tag{22}$$

$$\%\ \text{exhaust} = 1 - \%\ \text{work} - \%\ \text{heat loss} \tag{23}$$

where W_{tot} is the total amount of work done during the cycle, and Q_{tot} is the total heat transfer from the cylinder gases to the wall, and HVL is the lower heating value of the fuel.

Finally, the summary provides an estimate of the engine's exhaust temperature. Although the actual properties of the mass leaving the exhaust port are time varying, the value of temperature that provides the same average enthalpy transfer as the time-varying flow is determined by solving the following equation iteratively:

$$h(T_{exh}, P_{exh}, \phi_{exh}) - \frac{\int h_{out} dm_{out}}{\int dm_{out}} = 0 \tag{24}$$

where

T_{exh} exhaust temperature (unknown)

P_{exh} exhaust manifold pressure

ϕ_{exh} equivalence ratio in exhaust

This provides a mass-averaged exhaust temperature.

SAMPLE CASE

As a sample case, consider a two-stroke diesel engine with the following characteristics:

opposed piston design

0.00022 lbm fuel injected per stroke

150 psia intake pressure

885 °R intake air temperature

137.5 psia exhaust pressure

3.101 in. bore by 2.940 in. stroke

7.168 in. connecting rod length

9.1712 compression ratio

intake port opens at 125 °ATDC, occupies 55% of cylinder bore, flow coefficient of 0.8

exhaust port opens at 100 °ATDC, occupies 55% of cylinder bore, flow coefficient of 0.8

The program below calls the subroutine DSL2 to calculate the performance of the two-stroke diesel engine. The intake pressure and temperature and the exhaust pressure are input through the subroutine argument list. The remaining data are provided in the input data set named DSL.INP. This data set is provided in table II. The complete output file is also printed below.

```
C - SAMPLE PROGRAM
C
      PI=150.
      TI=885.
      PEXH=137.5
C
      CALL DSL2(TI,PI,PEXH,TEXH,XM)
C
      STOP
      END
```

Complete Output File

```
BORE                               3.101 INCHES
STROKE                             2.940 INCHES
CONNECTING ROD                     7.168 INCHES
GEOMETRIC COMPRESSION RATIO        9.17
EFFECTIVE COMPRESSION RATIO        6.21
SWEPT VOLUME                      44.409 CUBIC INCHES
ENGINE SPEED                    6122. RPM
MEAN PISTON SPEED               3000. FT/MIN
SUPPLY AIR PRESSURE              150.00 PSIA
SUPPLY AIR TEMPERATURE           885.00 DEG R
EXHAUST PRESSURE                 137.50 PSIA


PORT TIMING:   INTAKE OPEN       125.
               INTAKE CLOSE      235.
               EXHAUST OPEN      100.
               EXHAUST CLOSE     260.


FUEL FLOW RATE   0.0224 LBM/SEC     FUEL INJECTED/CYCLE   0.000220 LBM/CYCLE
AIR FLOW RATE    0.6544 LBM/SEC     AIR INDUCTED/CYCLE    0.006413 LBM/CYCLE


IMEP               314.7 PSI           IHP              216.1 HP
BMEP               286.7 PSI           BHP              196.9 HP
FMEP                28.0 PSI           FHP               19.2 HP

ISFC             0.3739 LBM/HP-HR
BSFC             0.4104 LBM/HP-HR

INDICATED THERMAL EFFICIENCY       0.3738
BRAKE THERMAL EFFICIENCY           0.3405
```

MAXIMUM PRESSURE	2784.4 PSIA	AT 368.8 DEG
MAXIMUM TEMPERATURE	4571.5 DEG R	AT 378.0 DEG
MAXIMUM RATE OF PRESS RISE	82.81 PSI/DEG	AT 355.0 DEG

PURITY	0.7343
SCAVENGING EFFICIENCY	0.6965
TRAPPING EFFICIENCY	0.5941
CHARGING EFFICIENCY	0.3241
DELIVERY RATIO	0.5455
RESIDUAL FRACTION	0.3164

EQUIVALENCE RATIO DURING COMPRESSION	0.2528
EQUIVALENCE RATIO AT EVO	0.8304
EQUIVALENCE RATIO BASED ON FUEL AND AIR FLOW RATES (EXHAUST)	0.4932

% FUEL ENERGY TO HEAT LOSS	8.11 %
% FUEL ENERGY TO WORK	37.38 %
% FUEL ENERGY TO EXHAUST	54.52 %

EXHAUST TEMPERATURE 2057.5 DEG R

CA	PCYL	TCYL	PHICYL	MCYL	MDOTIN	MDOTOUT	BALANCE
105.	313.0	3068.1	0.8304	0.005692	0.000000	0.000040	0.00000
110.	276.0	2985.0	0.8304	0.005409	0.000000	0.000071	0.00000
115.	236.2	2885.8	0.8304	0.005000	0.000000	0.000090	0.00000
120.	197.9	2776.6	0.8304	0.004528	0.000000	0.000096	0.00000
125.	165.3	2669.0	0.8304	0.004075	0.000000	0.000082	0.00000
130.	142.8	2578.1	0.8271	0.003762	0.000012	0.000043	0.00000
135.	137.4	2512.4	0.8014	0.003818	0.000030	-0.000008	0.00000
140.	137.5	2446.6	0.7624	0.004022	0.000043	0.000000	0.00000
145.	137.6	2366.5	0.7162	0.004249	0.000055	0.000007	0.00000
150.	137.7	2279.5	0.6666	0.004496	0.000065	0.000014	0.00000
155.	137.9	2190.6	0.6168	0.004756	0.000073	0.000020	0.00000
160.	138.1	2103.2	0.5689	0.005021	0.000079	0.000026	0.00000
165.	138.3	2019.8	0.5241	0.005286	0.000084	0.000032	0.00000
170.	138.6	1941.8	0.4830	0.005542	0.000087	0.000037	0.00000
175.	138.8	1870.2	0.4459	0.005785	0.000089	0.000042	0.00000
180.	139.0	1805.2	0.4127	0.006010	0.000089	0.000046	0.00000
185.	139.2	1746.8	0.3833	0.006210	0.000087	0.000050	0.00000
190.	139.4	1694.7	0.3575	0.006383	0.000084	0.000053	0.00000
195.	139.6	1648.8	0.3350	0.006525	0.000080	0.000055	0.00000
200.	139.8	1608.7	0.3156	0.006632	0.000074	0.000056	0.00000
205.	140.0	1574.3	0.2990	0.006702	0.000067	0.000057	0.00000
210.	140.1	1545.1	0.2851	0.006733	0.000059	0.000056	0.00000
215.	140.3	1521.2	0.2737	0.006723	0.000049	0.000055	0.00000
220.	140.5	1502.4	0.2647	0.006672	0.000038	0.000053	0.00000
225.	140.7	1488.8	0.2582	0.006578	0.000027	0.000050	0.00000
230.	140.9	1480.5	0.2542	0.006441	0.000014	0.000045	0.00000
235.	141.1	1478.0	0.2528	0.006261	0.000000	0.000040	0.00000
240.	141.9	1480.1	0.2528	0.006070	0.000000	0.000037	0.00000
245.	143.8	1485.0	0.2528	0.005894	0.000000	0.000033	0.00000
250.	147.0	1493.3	0.2528	0.005739	0.000000	0.000028	0.00000

255.	152.4	1506.9	0.2528	0.005622	0.000000	0.000018	0.00000
260.	161.5	1529.1	0.2528	0.005574	0.000000	0.000000	0.00000
265.	174.3	1558.4	0.2528	0.005574	0.000000	0.000000	0.00000
270.	189.3	1590.8	0.2528	0.005574	0.000000	0.000000	0.00000
275.	207.1	1626.5	0.2528	0.005574	0.000000	0.000000	0.00000
280.	228.1	1665.9	0.2528	0.005574	0.000000	0.000000	0.00000
285.	253.3	1709.2	0.2528	0.005574	0.000000	0.000000	0.00000
290.	283.5	1756.8	0.2528	0.005574	0.000000	0.000000	0.00000
295.	320.0	1809.0	0.2528	0.005574	0.000000	0.000000	0.00000
300.	364.2	1866.2	0.2528	0.005574	0.000000	0.000000	0.00000
305.	417.9	1928.8	0.2528	0.005574	0.000000	0.000000	0.00000
310.	483.6	1996.9	0.2528	0.005574	0.000000	0.000000	0.00000
315.	563.9	2070.7	0.2528	0.005574	0.000000	0.000000	0.00000
320.	661.7	2149.8	0.2528	0.005574	0.000000	0.000000	0.00000
325.	780.0	2233.5	0.2528	0.005574	0.000000	0.000000	0.00000
330.	920.6	2320.5	0.2528	0.005574	0.000000	0.000000	0.00000
335.	1082.9	2408.2	0.2528	0.005574	0.000000	0.000000	0.00000
340.	1261.4	2492.7	0.2528	0.005574	0.000000	0.000000	0.00000
345.	1442.5	2568.6	0.2528	0.005574	0.000000	0.000000	0.00000
350.	1704.3	2788.6	0.2848	0.005586	0.000000	0.000000	0.00000
355.	2094.0	3230.4	0.3707	0.005619	0.000000	0.000000	0.00000
360.	2481.3	3730.8	0.4848	0.005662	0.000000	0.000000	0.00000
365.	2730.4	4151.8	0.5986	0.005706	0.000000	0.000000	0.00000
370.	2776.7	4426.4	0.6916	0.005741	0.000000	0.000000	0.00000
375.	2641.3	4553.0	0.7563	0.005766	0.000000	0.000000	0.00000
380.	2392.7	4565.5	0.7950	0.005781	0.000000	0.000000	0.00000
385.	2100.3	4504.7	0.8152	0.005788	0.000000	0.000000	0.00000
390.	1811.6	4403.1	0.8245	0.005792	0.000000	0.000000	0.00000
395.	1550.9	4282.9	0.8283	0.005793	0.000000	0.000000	0.00000
400.	1326.8	4157.7	0.8296	0.005794	0.000000	0.000000	0.00000
405.	1138.8	4034.6	0.8300	0.005794	0.000000	0.000000	0.00000
410.	983.1	3917.2	0.8302	0.005794	0.000000	0.000000	0.00000
415.	854.8	3807.0	0.8302	0.005794	0.000000	0.000000	0.00000
420.	748.9	3704.4	0.8302	0.005794	0.000000	0.000000	0.00000
425.	661.5	3609.6	0.8302	0.005794	0.000000	0.000000	0.00000
430.	588.9	3522.2	0.8302	0.005794	0.000000	0.000000	0.00000
435.	528.5	3441.8	0.8302	0.005794	0.000000	0.000000	0.00000
440.	477.8	3368.1	0.8302	0.005794	0.000000	0.000000	0.00000
445.	435.2	3300.5	0.8302	0.005794	0.000000	0.000000	0.00000
450.	399.2	3238.7	0.8302	0.005794	0.000000	0.000000	0.00000
455.	368.6	3182.3	0.8302	0.005794	0.000000	0.000000	0.00000
460.	342.6	3130.7	0.8302	0.005794	0.000000	0.000000	0.00000

CONCLUSION

The program described in this report can be used to simulate any two-stroke engine with either a standard piston-cylinder configuration or an opposed piston configuration. The program was developed to provide a way to predict the performance of the diesel core of a high-output combined-cycle diesel engine. The sample case presented is one example of a possible engine suitable for this use.

This program can be used to investigate the effects on engine performance of combustion timing, valve or port timing, heat transfer, and engine geometry. Gross engine parameters such as bore, stroke, connecting rod length, and compression ratio as well as engine operating parameters such as speed, load, and altitude can be easily varied. While exact prediction of engine characteristics is not possible, the program should accurately predict trends.

REFERENCES

1. Annand, W.J.D.: Heat Transfer in the Cylinders of Reciprocating Internal Combustion Engines. Proc. Inst. Mech. Eng., vol. 177, no. 36, 1963, pp. 973-996.

2. Olikara, C.; and Borman, G.L.: Computer Program for Calculating Properties of Equilibrium Combustion Products with Some Applications to I.C. Engines. SAE Paper 750468, 1975.

3. Watson, N.; Pilley, A.D.; and Marzouk, M.: A Combustion Correlation for Diesel Engine Simulation. Diesel Combustion and Emissions, SAE P-86, Society of Automotive Engineers, 1980, pp. 51-63.

4. Schweitzer, P.: Scavenging of Two-Stroke Cycle Diesel Engines. Macmillan, New York, 1949.

TABLE I. - INPUT VARIABLES

RPM	WFCY				
BORE	STROKE	CONROD	CR	MODE	
C1	C2	C3	C4	C5	
ANNND	THEAD	TPISTN	TSLEEV		
MEXH	EVO	EVC	MINT	AVO	AVC
NTEXH	CDEXH				
ALPHEX(1)	FEXH(1)				
ALPHEX(2)	FEXH(2)				
.	.				
.	.				
.	.				
ALPHEX(NTEXH)	FEXH(NTEXH)				
WIDTHI	CDINT				

TABLE II. - DSL.INP

6122.	0.00022				
3.101	2.940	7.168	9.1712	2	
1.0	345.	55.	0.0	3.5	
0.7	1460.	1460.	1460.		
0	100.	260.	0	125.	235.
0.55	0.8				
0.55	0.8				

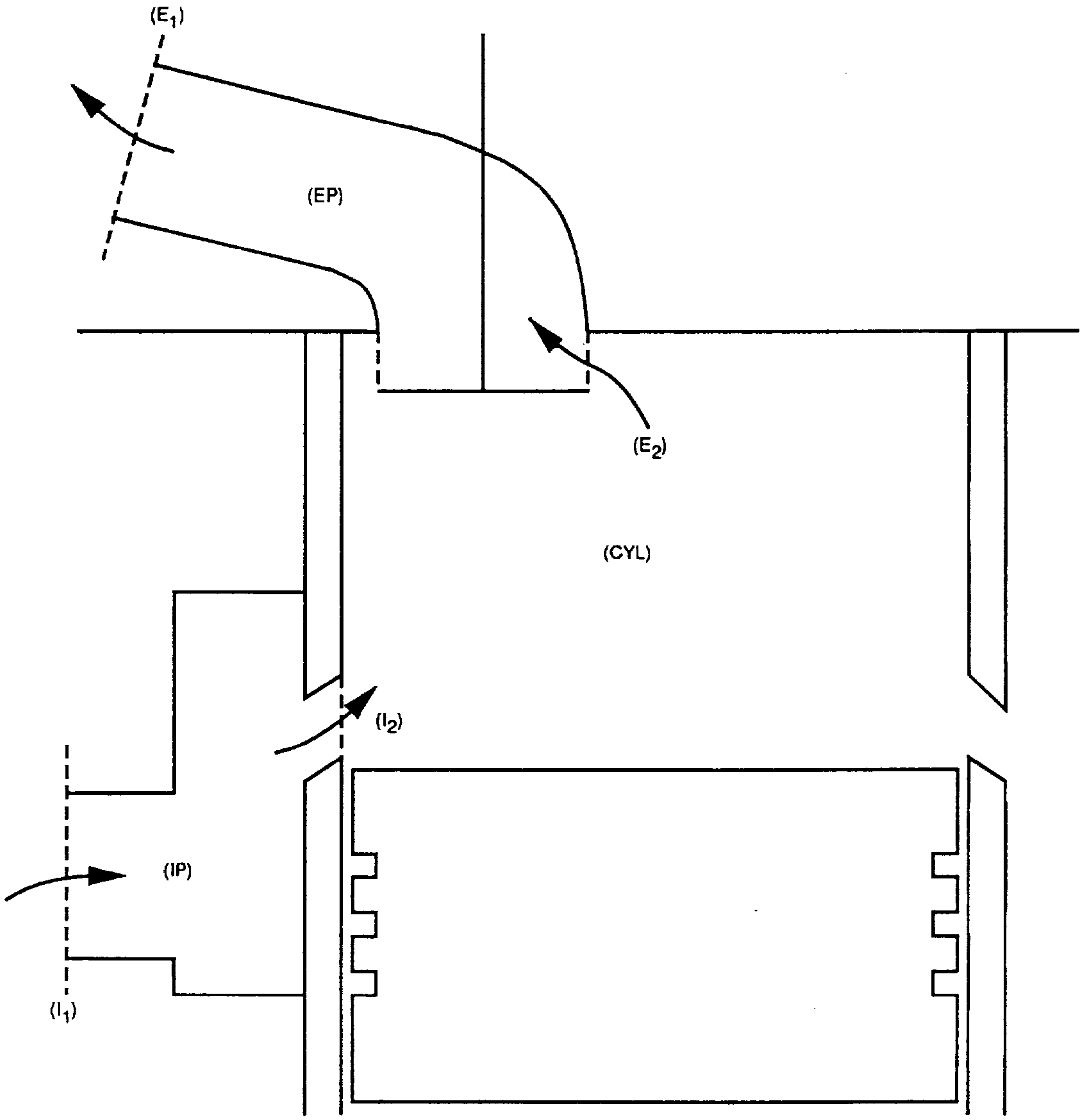

Figure 1. – System diagram.

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15. Supplementary Notes

Project Manager, Robert C. Bill, Propulsion Directorate, U.S. Army Aviation Research and Technology Activity—AVSCOM. Jon H. VanGerpen, Summer Faculty Fellow at Lewis Research Center.

16. Abstract

A computer program simulating a two-stroke diesel engine is developed and documented. The program is suitable for simulating the diesel core of a high-output combined-cycle diesel engine. The engine cylinder and the intake and exhaust ports are defined as independent thermodynamic systems and the mass and energy equations for these systems are developed. A single zone combustion model is used and perfect mixing during scavenging is assumed. The program input requirements and output results are discussed. A sample case is provided for an opposed piston, uniflow scavenged two-stroke diesel engine.

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